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Design trends and challenges in hydrogen direct injection (H_2DI) internal combustion engines – A review

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ABSTRACT

The hydrogen internal combustion engine (H_2-ICE) is proposed as a robust and viable solution to decarbonise the heavy-duty on- and off-road, as well as the light-duty automotive, sectors of the transportation markets and is therefore the subject of rapidly growing research interest. With the potential for engine performance improvement by controlling the internal mixture formation and avoiding combustion anomalies, hydrogen direct injection (H_2DI) is a promising combustion mode. Furthermore, the H_2-ICE poses an attractive proposition for original equipment manufacturers (OEMs) and their suppliers since the fundamental base engine design, components, and manufacturing processes are largely unchanged. Nevertheless, to deliver the highest thermal efficiency and zero-harm levels of tailpipe emissions, moderate adaptations are needed to the engine control, air path, fuel injection, and ignition systems. Therefore, in this article, critical design features, fuel-air mixing, combustion regimes, and exhaust after-treatment systems (EATS) for H_2DI engines are carefully assessed.

1. Introduction

To achieve a structural shift towards decarbonising the energy landscape, OEMs and engine development partners need technological breakthroughs to improve existing internal combustion engines for future low-emission vehicles. Under the 'Fit for 55' package, the European Council has set a target for carbon dioxide (CO_2) emissions reduction of at least 55% for cars and vans by 2030, compared to 1990 levels, and to attain net-zero CO_2 levels by 2050 - with all 27 European Union (EU) member states agreeing to this commitment [1]. In the USA, the Environmental Protection Agency (EPA) has proposed stringent multi-pollutant emissions standards for the sales of new light-duty and medium-duty vehicles between 2027 and 2032 [2]. For example, the EPA has pushed for an average target of 82 g/mile of CO_2 for the light-duty fleet in 2032, which represents a 56% reduction in greenhouse gas emissions standards compared to 2026 model year vehicles. One widely proposed solution to meet these ambitious targets, due in part to their legislated 'zero emissions' status, is battery electric vehicles (BEVs). Full electrification of the fleet, however, is challenging due to issues, both perceived and real, related to limited driving range, charging infrastructure, region-specific availability of fully renewable electricity generation, demand constraints, and concerns over the mining and recycling of rare earth and precious metals [3–6]. For light-duty applications, electrification is a good alternative to the internal combustion engine; however, its true carbon intensity is not zero. Additionally, the freshwater eutrophication and ecotoxicity effects of using BEVs are higher than internal combustion engines due to their emissions to water when electricity is produced through coal mining, thereby, the need for low-carbon electricity generation is crucial [7,8]. For road vehicles, the non-exhaust particulates from tyre and road surface wear, particularly in the range of 10 and 2.5 μm (PM_{10} and $PM_{2.5}$), are higher for BEVs due to their being heavier (with respect to like-for-like vehicles) and having higher instantaneous torque than internal combustion engine vehicles [9].

Based on a recent projection, around 75% out of 1.6 billion global passenger cars on the road in 2040 will be powered either by an ICE either solely or as part of a hybrid powertrain [10]. The continuous improvement of engine efficiency is therefore of paramount importance

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in the near term with a shift towards the use of low-carbon or zero-carbon fuels in the long term [11,12]. Hybrid powertrains with low or high levels of electrification combined with simpler or advanced ICEs also offer potential advantages in many markets [13–15]. In short, technology-agnostic approaches should be adopted to design suitable powertrains for a given region and application [16,17].

The development of a hydrogen economy in the energy and transportation sectors can assist in achieving climate change mitigation targets. The advent of extraction methods to produce 'blue' hydrogen through natural gas or coal gasification coupled with carbon capture and storage (CCS) [18,19] 'green' hydrogen via water electrolysis through renewable energy electricity generation [20,21], and 'pink' hydrogen using nuclear energy for electrolysis [22] makes hydrogen utilisation technically and economically feasible [23]. From a well-to-wheel and life-cycle CO_2-free mobility standpoint, hydrogen has the potential to play a vital role in either fuel cells for electric powertrains (H_2-FCEV) or internal combustion engines (H_2-ICE). From the context of the total cost of ownership (TCO) and CO_2 equivalency, hybridised H_2-ICE powertrains offer a medium and long-term alternative to the H_2-FCEV and BEVs for light-duty commercial vehicles [24]. Whilst similar storage tanks are required for both H_2-FCEV and H_2-ICE, the H_2-FCEV incurs higher production costs due to the requirement for precious metals, high-performance auxiliary air compressor units, large battery packs, and an extremely capable low-temperature cooling circuit and associated radiators for high power and low-temperature heat rejection. Since the overall powertrain system efficiency of an H_2-FCEV can be similar to that of an H_2-ICE [25], H_2-ICE powertrains are a viable option for passenger car applications where higher power density, increased efficiency at high loads, and lower capital expenditure (CapEx) are key requirements [26,27]. Here, the H_2-ICE should not be considered as a competitor to H_2-FCEV technology, but rather as a complement, to exploit established supply chains and existing production infrastructure.

Over the years, significant research and development activities have been channelled to redesign conventional powertrains for H_2-ICE by addressing specific challenges related to air-path, fuel injection, mixture formation, ignition, combustion, and EATS. The fuel injection strategy, either port fuel injection (PFI) or direct injection (DI), and the detail of the fuel injection system as a whole have been shown to be critical to the successful implementation of H_2-ICE. PFI is a potentially cost-effective and straightforward solution but requires further technology modifications to be competitive. PFI typically leads to reductions in volumetric efficiency due to the displacement of inlet air with hydrogen, which leads to reduced power density. This can be compensated for either with a powerful boosting system [28,29] or by injecting cryogenic hydrogen in the intake port at extremely low temperatures [30]. Without these technologies, the penalties of a PFI hydrogen strategy can be severe. BMW developed a 'mono-fuel' and a bi-fuel (hydrogen and gasoline) hydrogen engine in their 7 Series vehicle based on a V12 spark-ignition production unit using PFI and no intake air boosting. This led to a 41.6% decrease in specific power output from the engine for hydrogen operation compared to gasoline operation [31,32]. The project was subsequently discontinued due to the engine's relatively poor fuel efficiency and the fact that it was not considered to be a zero-emission vehicle by the US EPA (due to the burning of small amounts of lubricating oil in the combustion chamber [9,33]). Ford also developed the PFI-based 'P2000H2ICE' from an existing Zetec 2.0 L gasoline engine and reported, relative to the gasoline variant, a power reduction of up to 50% at high engine speeds and 35% at low-medium engine speeds [34–37], similarly attributed to the displacement of combustion air by the injected hydrogen gas. In parallel, research has been conducted on retrofitting or adapting incremental changes to the existing compression-ignition production infrastructure with dual-fuel capability [38–43]; however, refuelling two fuel tanks can be practically unviable.

Hydrogen Direct Injection (H_2DI) can provide higher specific power output, improved efficiency, and better transient response compared to PFI due to the reduced pumping work and lower demands on the boosting system [44,45]. To control the occurrence of pre-ignition and backfiring for PFI, the injection window during early-intake stroke needs to be adjusted such that no residual hydrogen is left in the intake manifold after the intake valve closing. These drawbacks can be alleviated using H_2DI fuel systems, which can also allow a certain level of scavenging without causing hydrogen slip.

Early direct injection requires medium injection pressure (i.e., up to 40 bar) while late-cycle direct injection requires high injection pressure (i.e., 50–300 bar) such that the injection pressure is sufficiently higher than the maximum in-cylinder pressure and sufficient hydrogen can be introduced in a shorter period of time [46–48]. Compared to late injections with limited fuel-air mixing time, early injections result in lower NO_x emissions due to higher mixture homogeneity and low combustion temperatures. However, this NO_x trend is highly reliant on engine load and global air-to-fuel equivalence ratio [44]. In general, DI can enhance charge motion and turbulence to increase flame speed and improve mixture preparation, which is beneficial for lean hydrogen combustion, as will be discussed in the following section. Despite a relatively short fuel-air mixture preparation window (i.e., ~4–20 ms at engine speeds between 1000 and 5000 rpm), DI presents opportunities for further optimisation of engine control, efficiency, and emissions relative to PFI [49–52]. In 2009, BMW successfully demonstrated an indicated thermal efficiency of 45% with a 200-bar fuel injection system using DI operation [53] strongly outperforming the company's previous PFI approach.

The low density of hydrogen presents technical and economic challenges for its storage and transportation, requiring either high pressures of 350–700 bar in gaseous form, or temperatures below −253 °C in liquid form [54]. An investigated approach to address these concerns is an onboard fuel reforming of liquid fuels with recuperation of the waste heat from the fuel reforming process [55,56]. Further challenges associated with mass production include:

- the particulate formation resulting from the combustion of in-cylinder lubricating oil,
- control of NO_x emissions,
- accumulation of hydrogen and water vapour in the crankcase,
- durability of the ignition and injection system,
- demands on the turbocharging system to supply sufficient air for lean combustion.

In summary, significant research and development activities are necessary to achieve high brake thermal efficiency targets with minimal NO_x emissions to compare favourably with the latest generation of gasoline and diesel engines. Whilst previous review articles have discussed certain aspects of hydrogen engine fundamentals and technology [3,25,44,50,57–59] as well as a recent review from Nguyen and Turner [60] depicted the current state of hydrogen viability for light-duty engines considering the whole hydrogen economy, the present article focuses particularly on the H_2DI engine architecture that have been highlighted in the literature over the past decade. Overall, the review article aims to improve the understanding of the key design trends, challenges, and optimisation needs for H_2-ICE and emphasise an additional requirement for continued research and development in the field.

2. Air-to-fuel equivalence ratio (λ)

Hydrogen exhibits a wide flammability range (i.e., $\lambda = 0.14$–10 versus 0.26–1.51 for iso-octane at 300 K and 1 atm [44]) which allows the operation of H_2-ICEs at lean and ultra-lean conditions. In this paper, lean combustion is defined with dilution limits in terms of relative air-to-fuel ratio between 1 and 2.5 (i.e., $1 < \lambda < 2.5$) whilst ultra-lean combustion is described by dilution levels over 2.5 (i.e., $\lambda > 2.5$), which is in line with recent studies [61,62]. The lean operation can mitigate abnormal combustion events and reduce NO_x to near-zero levels; however, exhaust after-treatment systems (EATS) are often required for transients and conditions where the mixture preparation is not sufficient to maintain $\lambda > 2$ across the entire cylinder. Ultra-lean combustion can eliminate the requirement for after-treatment but places increased demands on the charging system [44,61].

It should be noted that challenges for an accurate estimation of the air-fuel ratio for H_2-ICE need to be addressed particularly during ultra-lean and transient conditions [35]. This is critical as the transition region from knock and NO_x-limited to misfire-limited operation can be quite narrow, i.e., ~0.3 λ [29]. For test bench operation, a mass flow-based λ calculation has the potential to be more accurate than a recalibrated or recharacterised lambda sensor (the O_2 concentration to λ curve being replaced with a hydrogen-specific curve), given the expected high accuracy from the flow rate measurement equipment [63,64]. An alternative method for λ calculation can be based on exhaust gas analysis using the Simons method; however, a reliable hydrogen analyser is needed to account for moderate levels of unburnt hydrogen (H_2-slip) [64]. This presents a challenge for an accurate on-vehicle lambda measurement.

3. Engine design areas

3.1. Piston

The wide flammability limits and fast burning velocity of hydrogen [44] make it an attractive fuel particularly when operated in a lean combustion regime. Studies suggest that lean hydrogen mixtures are less prone to end-gas autoignition than hydrocarbon fuels [25], which allows for a higher compression ratio piston to enable greater thermal efficiency. This has been demonstrated by Laget et al. [65], who achieved a compression ratio of 16:1 at lambda 2.5 without abnormal combustion. Piston crown designs can be adjusted to reduce the clearance volume at the top dead centre (TDC) and thus increase the compression ratio. Enhancements to tumble and turbulent kinetic energy (TKE) of in-cylinder flows have also been demonstrated with simple bowl-shaped piston crowns [66]. Researchers at the University of Orleans have performed a comparison between flat and bowl-shaped piston geometries on a modified diesel engine [67], their conclusions indicate that the choice of piston design interacts with a number of operating parameters including injection and spark timing, with likely dependencies on the design of the combustion chamber and ports.

Material choice and cooling method are critical factors for the temperature field of the piston crown, and therefore the occurrence of hotspots that can trigger premature ignition events. For example, when compared with diesel steel pistons which showed hotspots at the localised positions of diesel spray, hydrogen steel pistons with a shallow bowl were found to have hotspot regions only at the bottom of the bowl – which the authors attributed to a larger distance from the cooling gallery [68]. If the material is changed from steel to aluminum along with a secondary underside cooling jet, hotspot regions, and therefore the risk of pre-ignition, can be further reduced due to the higher thermal conductivity and shorter cooling gallery distances [68–70]. Note that the degradation phenomenon for the impact of long-term exposure to aluminum and its alloys in a hydrogen environment is not clearly understood [71]. Moreover, for aluminum pistons, it must be accepted that heat transfer losses are higher than for steel pistons which can lead to lower thermal efficiency [72].

To lower the blow-by losses, reduce the friction losses through controlled oil consumption, and decrease the entrapped hydrogen emissions, which are normally associated with high peak pressures and thermal loads, the piston ring package should be optimised. This includes proper design of rings, grooves, top ring land crevice volume, and piston skirt to wall clearance. Coatings of the top oil ring have also shown promise, including hard ionised coating [34] and chrome ceramic coating (CKS) as used for typical diesel engines [31] to ameliorate the burn-residue safety.

3.2. Intake port and valve

Low charge motion (i.e., tumble ratio) can contribute to reduced combustion speed, high NO_x emissions due to decreased mixture homogeneity, as well as increased occurrence of pre-ignition and knock [73]. To increase the flame propagation speed and achieve a faster heat release rate (HRR), enhanced turbulence within the mixtures is essential [74–76]. With a faster burn rate, particularly close to stoichiometry, the need for high charge motion can be reduced. However, for extreme lean conditions, the charge motion requires significant improvement for good mixture homogeneity and increased HRR whilst maintaining acceptable flow performance (i.e., flow coefficient).

In general, the tumble ratio and flow coefficient follow trade-off characteristics and these parameters could limit maximum engine power, gas exchange losses, and/or further burden the intake air boosting system if they remain unoptimised. Some methods to increase the tumble ratio include installing an additional tumble flap [77], a charge motion control valve (CMCV) upstream of the intake port [34] and via variable valve timings [78] but these have not been shown to have a significant benefit on tumble generation.

Alternative methods to achieve a high tumble ratio without compromising the flow coefficient significantly include modifications to the intake port features such as the lower part of the valve cross-section (A_T, A_V, and θ) [79,80], fish belly shape at the port floor (i.e., a downside offset to the centreline of the intake port to steer the airflow through the upper valve area), and masking design (H_M) on the outside of the intake valves [81], as shown in Fig. 1. While a higher included angle between the valves (θ_1) can enhance tumble flow, as the flow direction during the intake valve opening is directed towards the exhaust valves, a reverse-tumble flow can also be created [82,83], which indicates that θ_1 should be minimised. Moreover, for a pre-determined bore-to-stroke ratio, A_V remains fixed and a reduced masking height (H_M) could result in a higher tumble ratio without compromising the flow coefficient as identified by Toyota, using a 2.5L gasoline engine [84]. For H_2-ICE, considering the high amount of airflow required to achieve lean conditions, both the flow coefficient and tumble ratio need to be optimised, similar to gasoline engines. Although for H_2-ICE, specific intake port design features were not reported [85]; however, the study has demonstrated that a high tumble port results in 25% improved homogenization and mixing quality compared to a good cylinder filling port. With increased tumble intensity i.e., 2.5 times higher tumble than a low tumble intake port, a 69% reduction in NO_x emissions was observed, along with a 5.5% increase in thermal efficiency, operating at 3000 rpm and 10 bar IMEP [65].

For high-performance gasoline engines, sodium-filled exhaust valves are typically used to reduce the exhaust valve surface temperatures in the combustion chamber as the heat is dissipated from the valve head through the sodium core to the valve guide. This can be synergised with H_2-ICE applications to reduce the potential of local hot-spot regions, thus lowering the occurrence of pre-ignition [86,87]. The heat transfer and therefore specific fuel consumption can also be reduced either by polishing the valve head to a mirror finish [88] or using thermal swing coatings [89]. To minimize the wear from valves and valve seat inserts due to the low lubricity of hydrogen, these can be made of cast steel with a hard surface coating [34].

3.3. Combustion chamber

Previous studies have derived the combustion chamber design from either a diesel or a gasoline base engine. This choice then naturally imposes specific packaging constraints for the DI injector and spark plug. For instance, a central injector location is needed for swirl air charge motion-based diesel engine with a flat-cylinder head which confines the spark plug location to a lateral position. A pre-chamber [64,90,91] or multiple spark plug configuration [46] can be considered to resolve this issue; however, these alternatives can be an additional

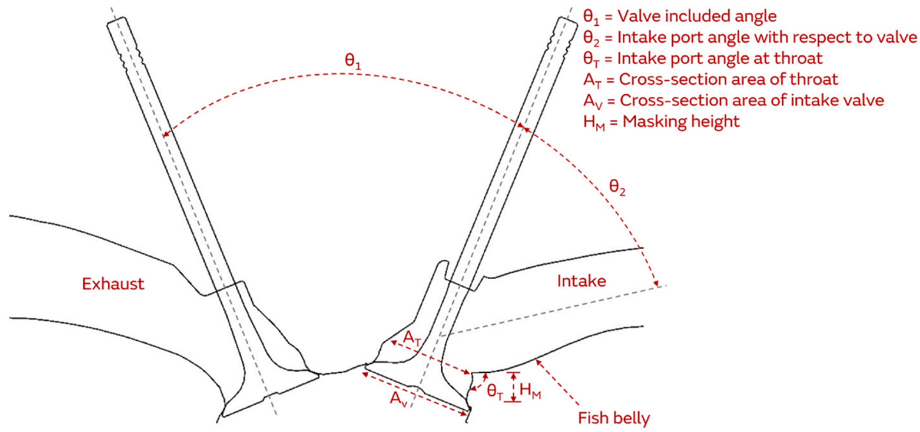

Fig. 1. Critical high-tumble intake port features derived from JLR gasoline engine.

pre-ignition source. In comparison, a tumble charge motion-based gasoline (e.g., GDI) engine with a pent-roof fire deck has the potential to generate better air-fuel mixing quality with both central and lateral injector configurations [46,92]. A recent study found that a combination of central spark plug location (to reduce the flame path), lateral-mounted injector between intake and exhaust valves and swirl-generating intake ducts provide an optimal mixing and combustion process [93]. This is because the hydrogen jet momentum overlays with the swirl motion during the intake stroke to cause strong in-cylinder tumble levels. Some researchers have proposed low turbulence-based combustion chambers such as pancake or disc chambers for the benefits of higher engine efficiency, decreased heat losses, and reduced abnormal combustion, particularly at stoichiometric engine operation [87,94,95].

Hydrogen combustion typically exhibits higher heat transfer losses relative to hydrocarbon-based fuels [96] due to its shorter quenching distance (0.64 versus 2.84 mm at $\phi = 1$ [25]) and higher flame temperatures [44,97], as illustrated in Fig. 2(a). To reduce the thermal loading, one promising countermeasure is the application of thermal swing coatings (TSC) on the in-cylinder components (piston crown, valves, exhaust port, combustion chamber roof) as it can result in an insulated layer through the combination of low thermal conductivity and low heat capacity of the coating [98–100]. Compared with an uncoated standard piston, a piston with TSC can cause the wall temperature to closely follow the bulk gas temperature, thereby showcasing the potential for improved thermal efficiency, particularly at low loads and lean operation [70]. Coating materials such as aluminium alloy, alumina, yttria-stabilised zirconia (YSZ), ceramics in the MgO–Al_2O_3–SiO_2 system, Keronite plasma electrolytic oxidation (PEO), and other commercial coatings have been tested for diesel and gasoline engines [101,102–105], the thermal properties of which are illustrated in Fig. 2(b); however, further research is needed with H_2-ICE.

3.4. Hydrogen direct injection (DI) injectors

To accurately meter the hydrogen injected mass, a choked flow (i.e., $Ma = 1$) through the injector nozzle is desirable as the flow rate is only reliant on injection duration, pressure, and needle lift. For hydrogen, the critical pressure ratio is ~0.53 at a polytropic coefficient of ~1.4 [44,106], meaning that the injection pressure should be greater than the in-cylinder back pressure by a factor of 2. The reasons that make hydrogen a challenging gaseous fuel compared to conventional liquid fuels are as follows:

- Flow rates: Due to the low volumetric energy density of gaseous hydrogen (i.e., 0.08 versus 692 kg/m^3 for iso-octane at 300 K and 1 atm [44]), the injectors require high volumetric flow rates with sonic flow conditions to satisfy the engine power demand.
- Leakage: Due to the high mass diffusivity of hydrogen (i.e., 0.61 versus ~0.07 cm^2/s for iso-octane at 300 K and 1 atm [44]), specific consideration is required for needle-seat sealing and bouncing.
- Lubrication: Hydrogen has low lubricity and viscosity which can compromise the injector's wear and durability [107,108].
- Dampening: The opening and closing profile of the injector needle needs to be smooth such that high-impact velocities and associated resonance effects can be reduced [46].
- Packaging: The low density of hydrogen also results in larger-sized injectors, particularly low-pressure injectors [109], which can be challenging for passenger car applications.
- Embrittlement: For mass production-based design, the right material selection of injector sub-components (e.g., austenitic and ferritic steels are preferred options over martensitic steels) is crucial to

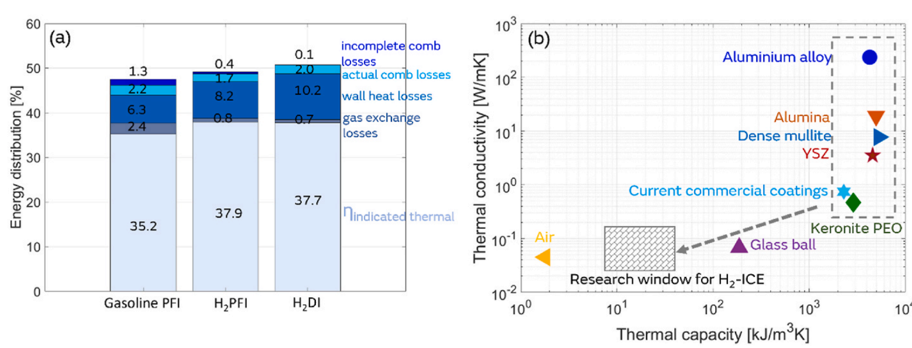

Fig. 2. Energy distribution for different engine configurations at 6 bar IMEP and 2000 rpm, adapted from Ref. [44], license number: 5805241192447 (a) and thermodynamic properties of the coating materials, reproduced from Ref. [101], license ID: 1492802-1 (b).

increase its durability without the need for an auxiliary oil supply [48,110].

A particularly important design parameter for hydrogen injectors is the needle opening direction, i.e., inward versus outward as shown in Fig. 3. Outward opening control injectors for hydrogen applications have gained much recent attention and are being actively researched worldwide [106,111–116]. Compared to inward-opening designs, outward-opening injectors typically employ larger nozzle cross-sectional areas, thus helping to alleviate the hydrogen flow rate concerns. Outward opening injectors also have excellent sealing and dampening capabilities [112,115] which can be beneficial for stratified (near-TDC) mixture preparation [117]. Significantly, for the outwardly opening injector approach, the high diffusivity of H_2 can favour the use of single-hole injectors with high discharge coefficients over more restrictive multi-hole injectors. This is because, for multi-hole injectors, a sharp change in flow direction occurs causing an unstable cavitation region to form downstream of the hole axis while the cavitation happens on the wall of the hole for single-hole injectors [118]. Further investigations on the jet cone structure have revealed its importance for both jet penetration and turbulence enhancement, with solid cones being beneficial for both [119], which also has implications for the hole design.

Of the major fuel injection equipment (FIE) suppliers, BorgWarner (now PHINIA Inc.) has developed medium-pressure outwardly-opening solenoid hydrogen injectors with a maximum flow rate of 10–15 g/s at 40 bar gas pressure [46,47]. The injection system has been tested in various thermodynamic engine configurations, and constant volume chambers [48,65,111,120–122], however, due to the low lubricating properties of hydrogen, an additional oil of 25 mg per kg of H_2 needs to be fed into the injector through an oil distribution system [120,123] or micro-lubricator [48]. This clearly limits the real-world adoption of the injector in its current development maturity due to the introduction of oil in the combustion chamber (although in ppb levels) which can contribute to particulate emissions. PHINIA has recently developed a variant capable of 60 bar injection pressure at a static flow rate of 15 g/s, which was successfully tested in an AVL RACETECH 2-Litre, 4-cylinder engine [26]. PHINIA has also been developing an inward opening multi-hole solenoid injector with a maximum flow rate of 60–90 g/s at 300 bar supply pressure and claims that the injector has the potential to achieve 4–10% higher relative engine efficiency than their medium pressure injector [46]. With an increased injection flow rate, the high-pressure injector allows near-TDC injection timings within a short duration, thereby reducing the compression work undertaken by the engine. Higher injection pressures have also been reported to improve jet penetration and mixture preparation [124]. To maximise the vehicle range and efficiency benefits, the pressure and the corresponding injection window can be optimised such that it can only be used when sufficient pressure is available from the tank system (i.e., H_2 storage pressure of 350/700 bar), otherwise, the injection window can be shifted to earlier optimised injections with lower injection pressure. This effectively eliminates the need for onboard gas compressor whilst reducing the cost and system complexity [125].

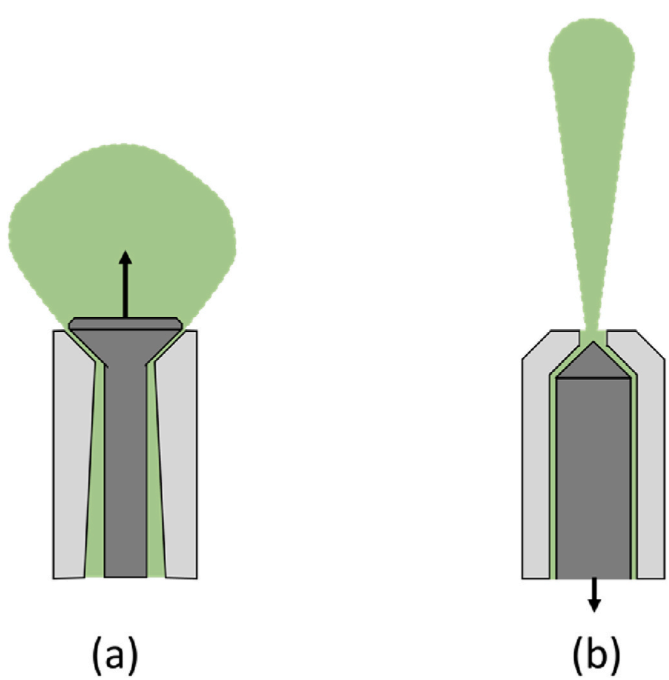

Fig. 3. (a) Outward-opening and (b) Inward-opening injector designs.

Comparisons between side and centrally mounted injector locations have shown high mixture homogeneity to be achievable with both configurations [46,121]. To further enhance the mixing quality, the jet shape can be controlled and targeted within the piston bowl via a deflector cap, the design of which can be optimised over the tip of the nozzle [47]. The deflector cap generates a volume downstream of the injector seat with the pressure kept above the in-cylinder pressure during the injection event. Note that these injectors were designed such that at the injector seat, sonic flow conditions are achieved, i.e., the injected mass flow rate remains independent of the downstream pressure until the critical pressure ratio is achieved. However, with the injector tip subjected to the hot combustion gases, the deflector cap can be a potential source of pre-ignition and therefore sufficient cooling and no injector protrusion into the combustion chamber are typically necessary [46]. This is further supported by the work of Bucherer et al. [126], who identified an effect on the knock limit of two different deflector cap designs where no difference in jet penetration was observed.

Another supplier, Bosch, has been primarily focused on developing low-pressure outward opening solenoid injectors with operating pressures of 20–40 bar without any additional lubricant supply (by separating the hydrogen flow from the injector) with expected series production in 2025/26 [92,93,127,128]. The needle lift curve for this configuration was found to be independent of the inlet pressure which could be beneficial to solve the issues of low injection quantity required during idling operation and quick purging when the vehicle is switched off [129]. The injectors can be adapted with or without customized jet guiding caps (JGC). It has been demonstrated that an optimised injector pocket design coupled with an injector recess position can lead to a high uniformity index (i.e., the relevance of tumble levels on the mixture formation) whilst keeping the hot jets away from the spark plug and exhaust valves [85]. An additional degree of flexibility for the jet targeting and mixture homogeneity can be provided using JGC; however, with a limitation of incomplete scavenging from the JGC region leading to rich mixtures around the injector tip [130]. Another study examined an electro-hydraulically actuated solenoid injector developed by Tokyo City University using diesel as the working oil with a maximum injection pressure of 200 bar for the durability of around 700 h and installed in a multi-cylinder engine [131,132]. They showed the advantages of improved lubrication, no hydrogen gas leakage at the needle valve, and multiple injection capability with high response time.

Whilst solenoid-driven injectors are cost-effective, they can suffer from limited needle motion control capabilities [108,133]. On the other hand, piezoelectric injectors have better needle control with minimal bouncing effect and faster response time via the assistance of soft opening and landing pulse profiles [134]. Through injector profile shaping (sometimes referred to as digital rate shaping, common in diesel engines [135]), the injection momentum of jets can be effectively controlled, and close-coupled multiple injections can be employed [107]. The application of piezo injectors in H_2-ICE engines can be extended for high-pressure injection, as developed by Westport Innovations Inc. and Continental Deka [112,136]. However, the current knowledge of piezoelectric actuators, in particular, the actuator materials in a hydrogen environment is quite limited [137].

3.5. Ignition system

The high autoignition temperature of hydrogen (i.e., ~853 K versus ~523 K for diesel [53]), suggests that burning solely hydrogen in a compression ignition engine without the use of a positive ignition source is difficult [138]. Although previous studies have reported the operation of hydrogen compression ignition engines including large-bore stationary engines [139], heavy-duty truck engines with pre-chamber assembly [140], and light-duty engines with a spark plug and/or a glow plug [53], these studies are beyond the scope of the work.

Despite its high autoignition temperature, hydrogen also exhibits a low ignition energy [141] which presents a risk of abnormal combustion, especially in the absence of a combustion moderator. High heat range or cold-rated spark plugs are required to prevent the spark plug temperature from exceeding the fuel's autoignition limit [142]. Compared to a hot spark plug, a cold-rated spark plug has a short and thick insulator nose which dissipates the heat rapidly and thereby helping to prevent pre-ignition. Spark plug electrode materials must be carefully chosen. Platinum electrodes may act as a catalyst between hydrogen and oxygen reactions and should therefore be avoided, while nickel electrodes suffer from high erosion rates [34,143]. Iridium electrodes are suggested as an attractive option to platinum due to their high erosion resistance [144]. Finally, with respect to plug design, the volume of the spark plug pocket should be as small as possible to minimize the high-temperature residual gases trapped inside the pocket [145].

As for the remainder of the ignition system, the low electrical conductivity of H_2 compared to hydrocarbon combustion mixtures requires that the ignition coils need to be designed such that no residual charge is left after the firing operation as it can lead to pre-ignition. Backfire caused by pre-ignition can also be improved by either a high output voltage which can be addressed by increased secondary ignition voltage due to the lower ion concentration of the hydrogen flames [146,147] or decreased ignition voltage which can be tackled by a small spark plug gap. Electrode-less combustion such as laser ignition, which not only abates the formation of localized hotspot regions but also has the potential for burn rate improvement can also be considered [148–153]. Challenges still remain in the practical implementation of these technologies on production engines.

3.6. Intake air boosting system

Lean combustion at high loads can increase the boost pressure demand of an H_2-ICE up to 1.8 times higher than an equivalent output gasoline or diesel engine [46], mainly due to the high stoichiometric air-fuel ratio compared to gasoline. This presents a major challenge for a re-matched turbocharging system, particularly at low-end torque and maximum power output regions. A single-stage boosting system with an optimised combination of a highly efficient compressor and variable-geometry turbine (VGT) is unable to provide the required air mass flow because of compressor surge and the limited exhaust enthalpy of pre-mixed lean hydrogen combustion [46,120]. With a similar power density to a diesel engine, H_2-ICE results in about 100 °C lower exhaust gas temperature [92]. High power output can be attained using multi-stage VGT systems but with a penalty of increased exhaust backpressure, although this can be alleviated via electrification of the VGT system [46]. This finding was also supported by studies that implemented a 48V-driven electric supercharger (e-SC) coupled with either a single-stage [154] or a two-stage VGT system [85,120]. Considering the electrical power demand for an e-SC, the peak BTE can be reduced up to 2%; however, this e-SC power demand can be compensated by regenerative braking in a hybrid powertrain layout [85]. With an external supercharger, a 14.2% improvement in thermal efficiency can be achieved at a boost pressure of 40 kPa [155].

At increased boost pressures the compressor outlet temperature can reach up to 250 °C, which can deteriorate the compressor material [46]. The high water vapour content of the exhaust gas, relative to gasoline or diesel engines, also introduces the risk of corrosion of the turbine housing. The use of a low-pressure (LP) exhaust gas recirculation (EGR) system (i.e., exhaust gases fed upstream of the compressor) can add further risk of damage to the compressor wheel. Therefore, the design and material selection of the turbocharging system requires careful attention, for example use of alternative materials such as Titanium [46].

3.7. Crankcase ventilation and lubrication

H_2-ICE are prone to a significant amount of blow-by (even with an optimised piston ring package), due to the small size of hydrogen molecules, which can result in significant quantities of trapped hydrogen in the crankcase [156]. To combat this, an active crankcase ventilation (CCV) system is required to keep the total unburned hydrogen concentration in the crankcase below the lower explosion limit (i.e., 4 % v/v in air). This could be achieved via vacuum pump integration into a conventional diesel engine CCV system. For diluting the blow-by to low hydrogen concentration, an additional air intake can be created in the crankcase housing [157]. Chi et al. [120] estimated that 0.7–1.7% of injected hydrogen escapes to the crankcase at low engine speed past the piston ring pack. However, with a properly designed CCV system, crankcase gases can be recirculated back to the combustion chamber through the intake system and an oil separator resulting in a thermal efficiency improvement of up to 0.3–0.7% points [84].

Hydrogen combustion generates 2.8 times more water condensates by mass than gasoline combustion at a fixed torque requirement likely due to the higher dew point temperature of hydrogen combustion gas [145]. Therefore, the engine lubricating oil needs to have high resistance to emulsification (such as demulsifying or synthetic lubricants [44]) to avoid adverse effects on the rotor bearing stability [46]. In addition to oil emulsification, lubrication oil must also be checked for the impact of atomic hydrogen [157]. Experimental results have confirmed that engine oil lubricating quality and viscosity index deteriorated with substantially decreased concentrations of zincdialkyldithiophosphate (ZDDP) and esters in the presence of hydrogen [156]. With the breakdown of lubricants during the combustion process, hydrogen can be generated which through the passage to the crankcase can wear the piston rings [158]. To lower the friction coefficient, bio-lubricants such as modified microalgae oil (MMO) can be used; however, the highly acidic nature of MMO can present further challenges [159]. A recent study investigated a detailed in-cylinder tribological performance of H_2-ICE and showed that the use of degraded and lower viscous lubricant results in reduced friction losses but at the expense of load-carrying capacity [160]. A multi-stage oil separation system with a carbon filter can also be used to reduce the lubricating oil residuals [61].

3.8. Materials selection

Two major deterioration mechanisms that can impact H_2-ICE materials are hydrogen embrittlement (HE) and high-temperature hydrogen attack (HTHA). HE is caused due to the diffusion of hydrogen gas through the metal surface forming atomic hydrogen in the alloy (typically at a temperature lower than 150 °C), thereby resulting in reduced metal ductility and fatigue crack propagation [161,162]. HE occurs either due to metal corrosion or from another chemical reaction during metal processing with hydrogen diffusion into the material being comparatively faster than its molecular reaction [157]. Compared to austenitic steel, ferritic and martensitic steels are more critical for HE as the temperature and pressure levels have a major effect on the diffusion and effusion of hydrogen [46]. For HTHA, the degree of damage multiplies when the steel is subjected to a higher temperature (above 200

°C) and hydrogen partial pressures above the resistance threshold, both for an extended time [163]. Steel with high carbon content (e.g., carbon tool steel) is more susceptible to HTHA than that with low carbon content (e.g., mild steel). Engine components such as the intake manifold, throttle, and injection module are subjected to low-temperature hydrogen and are more susceptible to HE whereas combustion chamber parts including the injector, cylinder head, valves/valves seat, piston/piston rings, liner, and exhaust manifold are more at risk of HTHA. Preliminary research evaluating materials of the exhaust and turbocharger systems (i.e., either turbine after the combustion process or compressor through CCV and/or LP EGR systems) found H_2-ICE generated atomic hydrogen about 67% higher than for methane-based combustion [157]. Therefore, it is necessary to carry out a careful selection of materials for H_2-ICE components, an example of such a layout can be found in Azeem et al. [71].

4. Fuel mixing quality

In-cylinder mixture formation is critical to avoid locally leaner or richer regions which can contribute to either misfire or increased NO_x emissions respectively. To understand hydrogen jet morphology and visualize the fuel mixing quality, experiments have been primarily conducted in either constant volume chambers with optical access [106,112,129,164] or optical engines [49,111,165,166]. Various research studies have used helium as a substitute fuel for hydrogen to eliminate ignition risk since it has similar physicochemical properties of molecular weight, density, and viscosity [167,168]. Whilst the volume of both the jets was found to be similar, helium gas showed higher jet penetration than hydrogen jets [169]. The most common optical measuring techniques for injection/mixture formation include schlieren imaging [167,170], high-speed combustion luminosity [171], fuel-tracer laser-induced fluorescence (tracer-LIF) [111], electronically excited hydroxyl (OH*) chemiluminescence [49], OH-LIF [172], and particle image velocimetry (PIV) [173].

The interaction of jets with neighbouring jets, chamber walls, and bulk gas flow motion are complex phenomena that present a significant challenge for injector development, jet shape optimisation, combustion system design, and engine operating strategy. For a diesel-based H_2-ICE with swirl air charge motion and side (lateral) injector position, a tailored jet (i.e., with deflector cap) can result in improved mixing compared to a conical jet profile (i.e., without deflector cap) but localised fuel-rich zones can still be an issue [46]. The same study highlighted that with the combination of the central injector position and tailored jet profile, the mixture homogeneity was greatly improved by utilizing the benefits of swirl motion when injecting the fuel across the swirl plane. Not only narrow and wide single jet shapes but also multi-hole jet designs using clover deflector caps have been recently investigated [48]. The multi-hole clover jet design can be considered as a wide jet; however, the jets remain separate during the spray development process resulting in a high mixing and spray penetration rate. This means that the jet shape does not have to depend on the intake port design or piston geometry to promote the mixture formation but has the potential to create overall good mixture homogeneity, including near the spark plug, over a wide speed-load regime with an optimised deflector cap [46–48]. The injector recess into the cylinder head can also impact the jet shape and penetration distance. Multiple studies have demonstrated that flush-mounted or small recess position can cause an increased coanda effect, i.e., the jet collapses at the nozzle exit as it follows the cylinder head walls resulting in a smaller penetration distance [123,174]. At an optimal mounting location, the jet shape was seen to transition from an initial broad opening angle to a highly focused jet due to non-restricted air entrainment resulting in higher velocities along the jet axis [129,173]. A larger recess can also lead to reduced penetration distance, due to jet deformation from the increased shear loss as well as a higher volume of localised rich mixtures inside the injector [123]. With the visible emitted wavelength range of hydrogen, the rich mixture fractions can be easily visualised using tracer-LIF as illustrated in various fundamental investigations [111,175].

Fuel injection timing after intake valve closing (post-IVC) and injection pressure are important parameters for in-cylinder mixture distribution. As shown in Fig. 4, very advanced (early) injection during the intake and early compression strokes result in locally ultra-lean mixtures with a peak around $\lambda = 3.03$ that decrease combustion stability, damage the lubricant film, and increase compression (negative) work [176]. On the other hand, retarded (late) injection during the mid-compression stroke forms highly stratified, locally rich mixtures (i.e., $\lambda < 2$) that increase NO_x emissions. Fuel injection at 20 bar into high ambient in-cylinder pressure during the late-compression stroke can result in extreme fuel inhomogeneity with a peak close to zero fuel-to-air equivalence ratio and this can cause unstable combustion [112].

Further reducing the interval between injection and ignition, i.e., increased stratification of hydrogen at the time of ignition (injection timing of around 30°CA bTDC at 100 bar) under lean burn conditions, is a potential way of reducing pre-ignition and improving fuel economy [113]. This is mainly due to reduced cooling loss between the burned gas and cylinder walls [177,178]. Depending upon injection timing, the jet-wall interactions can enhance the mixture formation and combustion stability [166]. Additionally, different injection timings have been found to impact flame kernel development, i.e., either spark-initiated or autoignition-initiated flame [179]. A recent study showed that retarded injection timing of 100°CA bTDC can result in a higher tumble ratio and reduced fuel-rich mixture zones (below $\lambda = 2$) compared to advanced injection timings of 160°CA bTDC [123]. This is because advanced injection timing can lead to lean homogenous mixtures with slow flame speed [113]. H_2DI using a double injection strategy has also been demonstrated to improve the mixture formation and tumble strength by controlling the secondary fuel injection amount (up to 20%) due to increased mass inertia of the secondary jet [176].

The influence of fuel injection pressure (FIP) is closely coupled with the injection timing. For instance, at earlier injections, a higher FIP generates a richer central core mixture than the global equivalence ratio whereas a lower FIP produces locally rich regions in the injector vicinity. The absence of the atomisation process for hydrogen (because it is injected as a gas) compared to liquid fuels and FIPs of up to 300 bar during injection in the mid-compression stroke has resulted in only a small effect of FIP on thermal efficiency and mixture formation [176]. During late injection for both low and high FIPs, localised large-scale rich regions were produced; however, with significant differences in their spatial location [49]. Injecting late during the later parts of the compression stroke requires that the FIP be sufficiently higher than the in-cylinder pressure, which is currently a packaging challenge for hydrogen FIE suppliers, particularly for passenger car engines.

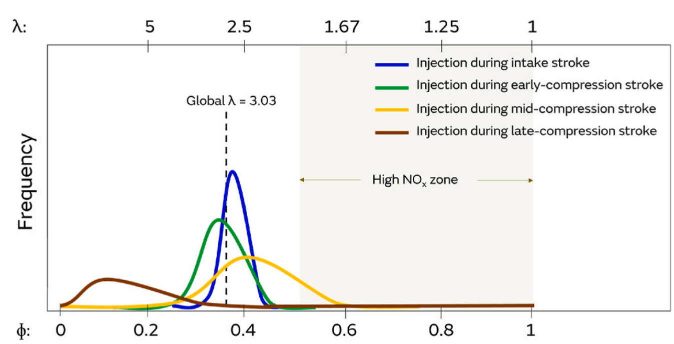

Fig. 4. Qualitative illustration of λ -distribution in the cylinder at ignition timing (around TDC) for different injection timings with the same global mixture λ (or ϕ) at an injection pressure of 20 bar. Based on results from Ref. [111], with permission from SAE International.

5. Combustion process

The criteria to achieve high thermal efficiency, sufficient power, low NO_x emissions, high combustion efficiency (i.e., low H_2-slip), and combustion stability with no abnormal combustion determine the requirements for mixture preparation and consequently the combustion process. Qualitative trade-off characteristics between these requirements for well-mixed H_2-air mixtures are shown in Fig. 5. NO_x emissions increase rapidly at $\lambda < 2$ and peak around $\lambda = \sim 1.3$ [44]. Lean combustion with $\lambda > 2$ has the potential to decrease NO_x emissions by keeping the combustion temperature below the threshold of thermal NO formation (i.e., 1800 K [180]). Ultra-lean operation with $\lambda > \sim 3.3$ is undesirable from an engine efficiency and very high boost pressure perspective due to reduced flame speed and stretched heat release rate, therefore negatively impacting the combustion phasing and combustion efficiency.

The stoichiometric operation can be favourable from the highest power output, torque, and moderate boost pressure perspective even though it increases NO_x emissions and the chances of abnormal combustion. To address these issues, Kapus et al. [181] recently demonstrated that stoichiometric operation coupled with a combustion moderator such as water injection and/or EGR (to handle irregular combustion) has the potential to achieve full load conditions. It was further highlighted in the same study that stable combustion with reasonable combustion speed and similar efficiency levels can be achieved with $\lambda = 1$ coupled with EGR addition rather than lean operation. This further permits the use of NO_x reduction catalysts, similar to three-way catalysts (TWC).

Lean operation coupled with EGR provides additional flexibility in reducing the frequency of abnormal combustion events along with a slightly lower boost pressure requirement compared to pure lean operation. This is because the minimum ignition energy of H_2/air mixtures increases rapidly when ϕ is decreased from the stoichiometric operating regime [182]. In addition, diluting air with an optimal EGR amount can lead to high specific power output, high efficiency, and low NO_x emissions [183,184]. Combustion anomalies in hydrogen engines include backfire, pre-ignition, and knock [185,186]. The risk of backfire is higher in PFI operation, where the combustible H_2/air mixture in the intake port can be ignited by hot gases or hotspots when the intake valve opens. Careful selection of intake valve timing and injection timing can eliminate the risk of backfire for DI operation. If premature combustion takes place after intake valves remain closed, it is known as pre-ignition or knock depending on its onset relative to spark timing [68,69]. The presence of local rich mixtures near the hot spot regions (e.g., spark plug, injector, exhaust valves) generally leads to pre-ignition. Better mixture quality and a dedicated combustion system with sufficient cooling capability of the cylinder head targeted at these critical hotspot areas can significantly reduce the frequency of pre-ignition [70,87].

With lean or ultra-lean combustion for H_2-ICE, only limited exhaust enthalpy is available for the turbine which is problematic for transient conditions in turbocharged applications. While a retarded injection timing can solve this concern, this can have a negative impact on engine efficiency. Another solution is a post-injection and post-combustion concept, i.e., injecting further hydrogen in the middle of the expansion stroke, as shown by Kufferath et al. [128]. With this approach, the boost pressure and torque are increased compared to a single injection combustion strategy, which leads to increased exhaust gas temperature and improved turbocharger response.

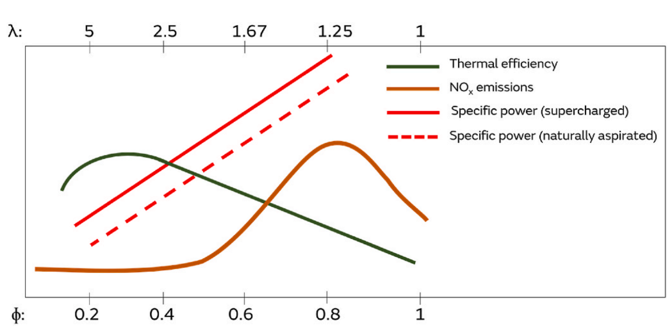

Fig. 5. The qualitative trade-off characteristics of H_2-ICE performance for homogenous mixtures at different global mixtures ϕ.

For stable lean combustion with homogenous mixture distribution, numerous studies reported high thermal efficiency of up to 45% at various speed-load mapped conditions [65,130,141,187–208] thanks to the high laminar flame speed, wide flammability limit, and high diffusivity of hydrogen [25]. Engine displacement, injection timings, fuel injection pressure (FIP), λ, peak indicated thermal efficiency (ITE), and peak brake thermal efficiency (BTE) from these studies are summarised in Fig. 6. 1D/0D simulation methods currently do not accurately capture the combustion and heat transfer phenomena of lean hydrogen combustion that have been observed experimentally [44,209] and so is an active area for further research.

6. Exhaust emissions and after-treatment technologies

The primary combustion product when burning hydrogen is water with each kg of hydrogen producing ~8.9 kg of water [29,63]. Therefore, the management of relatively large amounts of water vapour becomes important with solutions including controlling the mixture temperature with a restricted EGR rate [29], utilizing water recondensation from the exhaust line either into a port water injection system [181,210] or onto a heat exchanger such as an intercooler or engine radiator [211].

Numerous studies have found that carbon-based exhaust emissions from hydrogen combustion, i.e., HC, CO, CO_2, and PM are minimal with the combustion of small amounts of lubricating oil or the presence of background emissions in ambient air (PM, CO, CO_2, NO_x) drawn into the engine being the contributing sources [26,47,48,92,212]. However, an opposite trend of enhanced particle formation from hydrogen combustion compared to methane combustion, particularly at high loads has been recently reported [213,214]. This was explained by the low hydrogen flame quenching distance which caused increased lubricant evaporation, thereby resulting in entrainment of lubricant vapour in the in-cylinder bulk flow. In summary, particle emission characteristics from H_2DI might be strongly dependent on the combustion and injection system design, especially regarding the interactions of hydrogen jets with the oil films.

NO_x emissions are also a challenge for H_2-ICE because their formation depends primarily on the in-cylinder mixture (i.e., local air-fuel ratio) and the temperature of hydrogen flames. NO_x after-treatment ($DeNO_x$) for hydrogen engines can be adapted either from conventional diesel or gasoline engines which includes a three-way catalyst (TWC) for stoichiometric (traditionally gasoline) operation [215] while NO_x storage catalyst (NSC) [216], urea-based selective catalytic reduction (urea-SCR) [217], and H_2-SCR catalysts [218,219] are appropriate for lean-burn (traditionally diesel) operation. At stoichiometric conditions, SCR and NSC are inappropriate due to the lack of HC and CO as reducing agents. Therefore, a TWC is used for stoichiometric applications, but there is a challenge for applications while switching between stoichiometric and a lean burn mode, as the conversion efficiency of TWC drops significantly and rapidly at lean conditions [70]. New material development with hydrogen as the reductant is needed which is a challenge [220,221].

An overview of reported exhaust emissions of raw NO_x before SCR (bSCR), NO_x after SCR (aSCR), unburned hydrogen (uH_2), HC, and CO for various light-duty and heavy-duty applications are shown in Fig. 7. As mentioned earlier, for ultra-lean steady-state operation, the NO_x emissions are negligible, and theoretically, no EATS is required. However, during transient operation, dynamic NO_x emissions are formed which remains non-compliant with any EATS to date. In addition,

Ref.	Base engine	Engine capacity [L]	Cylinder volume [L]	Injection timing [°CA bTDC] (360 270 180 90 0)	FIP [bar] (0 100 200 300)	λ (0.5 1.5 2.5 3.5 4.5 5.5)	Peak ITE/BTE [%]
[52]	SI	0.49	0.49	130 – 30	45 – 150	1.0	37.0 (ITE)
[200]	SI	0.49	0.49	120 – 40	150	1.3 – 2.7	41.0 (ITE)
[113]	SI	0.50	0.50	38 – 26	100	2.5	35.0 (ITE)
[187]	SI	0.66	0.66	140 – 60	100	1.0 – 5.0	45.3 (ITE)
[188]	SI	0.66	0.66	140 – 60	100 – 140	1.0 – 5.0	47.1 (ITE)
[197]	SI	1.00	0.33	220 – 160	20	1.0 – 3.3	41.0 (BTE)
[194]	SI	1.48	0.49	300 – 50	50 – 150	2.3	42.5 (BTE)
[85]	SI	1.60	0.40	350 – 80	40	1.0 – 3.0	41.0 (BTE)
[145]	SI	1.60	0.53	210		1.0 – 3.0	25.0 (BTE)
[189]	SI	1.98	0.50		200	2.0 – 3.5	42.2 (BTE)
[198]	SI	1.99	0.50	160 – 50	60 – 100	1.1 – 3.3	-
[154]	SI	1.99	0.50	360	25 – 175	1.8 – 2.3	36.0 (BTE)
[195]	SI	2.00	0.50	310 – 160	70	1.3 – 1.4	39.1 (BTE)
[196]	SI	2.00	0.50		100	2.4	37.7 (BTE)
[190]	SI	2.00	0.50	200 – 50	40 – 200	1.0 – 4.0	42.0 (BTE)
[141]	SI	2.00	0.50	180 – 140	40 – 200	1.7 – 2.9	42.6 (BTE)
[127]	SI	2.00	0.50	300 – 40	100 – 100	1.8 – 3.0	45.0 (BTE)
[205]	SI	2.99	0.50	145 – 95		1.7 – 2.0	43.5 (BTE)
[201]	NG SI	1.30	1.30	150 – 25	110 – 170	1.7 – 2.3	42.0 (ITE)
[208]	NG CI	13.00	2.17		300		51.5 (BTE)
[204]	NG SI	7.70	1.28			2.0 – 5.0	42.0 (BTE)
[206]	CI	0.50	0.50	140		1.4 – 3.3	-
[199]	CI	1.84	1.84			1.8 – 4.4	48.3 (ITE)
[24]	CI	2.00	0.50		30	1.0 – 3.0	45.0 (BTE)
[47]	CI	2.00	0.50		20 – 40	1.6 – 2.6	37.5 (BTE)
[68]	CI	2.13	2.13	180	60 – 100	2.4 – 3.0	44.0 (ITE)
[121]	CI	2.17	0.54	120 – 100	25 – 35	1.6 – 2.7	34.0 (BTE)
[202]	CI	2.53	2.53	350 – 250	20	2.0 – 3.0	46.5 (ITE)
[207]	CI	12.80	2.13		250 – 300	1.4 – 2.9	42.0 (BTE)
[61]	CI	16.00	2.67	175		2.0 – 3.1	42.0 (BTE)
[203]	CI	16.80	2.80		22	2.0 – 4.0	47.0 (BTE)

Fig. 6. Engine peak thermal efficiency for lean combustion mode from various studies.

decreased exhaust gas temperature below the light-off temperatures of $DeNO_x$ either due to significant enleanment or during the cold start phase can lead to an inoperative EATS [6]. To solve this issue, one option could be a coupling of urea-SCR with an H_2-SCR system supported by Pt and Pd catalysts which has shown high conversion efficiency of up to 95% at around 150 °C [222–224]; however, further advances are needed to explore catalyst materials suitable at this temperature range. Another system layout of the EATS includes a combination of three-way NO_x storage catalysts (TWNSC) which has increased NO_x storage capability and SCR which can be used for both stoichiometric and lean operating modes [6,70]. For higher temperature limits, a combination of a TWC and urea-SCR can be used where urea-SCR can be fed passively through the secondary emissions of ammonia (NH_3) generated from the TWC, thereby avoiding the need for an additional urea tank. All these EATS layouts are shown in Fig. 8(a).

While various studies showed that the particulate emissions are several orders of magnitude lower for H_2-ICE than conventional gasoline or diesel engines [47,48], still to ensure that the future emission legislation limits are met (e.g., EU-7 and EPA 27), a particulate filter should be considered. Besides the emissions of NO_x, HC, carbon monoxide, and particulates, methane and nitrous oxide (N_2O) can also be produced, primarily over the catalysts [226]. Due to low concentrations of HC in the exhaust, methane emissions will be negligible. However, nitrous oxide can be formed either due to the oxidation of NO from the lean $DeNO_x$ system or due to the reduction of NO_x from the SCR catalyst [128].

Several studies identified that a simplified layout of a conventional diesel EATS is the most favourable solution for H_2-ICE as diesel engines operate on global lean mixtures with an excess of air in the exhaust. Such a layout includes a reduced size oxidation catalyst (compared to diesel) for unburnt hydrogen emissions, either a compact diesel particulate filter (DPF) or none at all for preventing particulates from lubricating oil combustion and urea-SCR system, a conventional SCR for reducing NO_x emissions, and an ammonia slip catalyst (ASC) for minimal NH_3 slip [46, 47,93,225], as shown in Fig. 8(b).

The potential pathways to simultaneously achieve high thermal efficiency, high power density, and low exhaust emissions are shown in Fig. 9. Lean or dilute combustion with direct injection, optimised boosting system, good thermal management including provisions for reduced abnormal combustion events and thermal swing coatings, and adaptable after-treatment systems can assure an advanced propulsion technology with zero harmful emissions. The diluent, be it air, EGR, or a combination of both, will increase the engines' need for low-loss gas exchange and a high-performance boosting system. Therefore, with an optimised design layout, H_2-DI presents a viable option for future SI engines.

Ref.	λ	NO_x bSCR			NO_x aSCR			uH_2		
	0.5 1.5 2.5 3.5 4.5 5.5	[min]	[max]	[unit]	[min]	[max]	[unit]	[min]	[max]	[unit]
[52]	1.0	5000.0	6000.0	ppm	-	-	-	1000.0	15000.0	ppm
[200]	1.3 2.7	8000.0	8000.0	ppm	-	-	-	-	-	-
[113]	2.5	200.0	400.0	ppm	-	-	-	-	-	-
[187]	1.0 5.0	0.1	1.6	g/kWh	-	-	-	-	-	-
[188]	1.0 5.0	100.0	350.0	ppm	-	-	-	-	-	-
[197]	1.0 3.3	10.0	4000.0	ppm	-	-	-	-	-	-
[194]	2.3	25.0	1481.0	ppm	-	-	-	-	-	-
[85]	1.0 3.0	0.2	10.0	g/kWh	-	-	-	250.0	3000.0	ppm
[145]	1.0 3.0	5000.0	5000.0	ppm	-	-	-	-	-	-
[189]	2.0 3.5	100.0	2999.0	ppm	2.0	138.0	ppm	2.0	138.0	ppm
[198]	1.1 3.3	-	-	-	-	-	-	-	-	-
[154]	1.8 2.3	100.0	1500.0	ppm	-	-	-	1000.0	17500.0	ppm
[195]	1.3 1.4	9.9	9.9	g/kWh	-	-	-	-	-	-
[196]	2.4	-	-	-	-	-	-	-	-	-
[190]	1.0 4.0	3642.0	3642.0	ppm	-	-	-	-	-	-
[141]	1.7 2.9	100.0	3000.0	ppm	20.0	20.0	ppm	20	20	ppm
[127]	1.8 3.0	1.0	9.0	g/kWh	-	-	-	-	-	-
[205]	1.7 2.0	2500.0	2500.0	ppm	-	-	-	-	-	-
[201]	1.7 2.3	1500.0	5800.0	ppm	-	-	-	3	10	%
[208]		6.2	12.2	g/kWh	-	-	-	0.004	0.015	g/kWh
[204]	2.0 5.0	0.5	3.2	g/kWh	0.05	0.19	g/kWh	-	-	-
[206]	1.4 3.3	2000.0	2000.0	ppm	-	-	-	-	-	-
[199]	1.8 4.4	0.0	0.9	g/kWh	-	-	-	2	6	g/kWh
[24]	1.0 3.0	8000.0	8000.0	ppm	-	-	-	-	-	-
[47]	1.6 2.6	1.0	9.0	g/kWh	-	-	-	-	-	-
[68]	2.4 3.0	0.3	2.0	g/kWh	-	-	-	-	-	-
[121]	1.6 2.7	1.0	27.0	g/kWh	-	-	-	-	-	-
[202]	2.0 3.0	4.0	4.0	g/kWh	-	-	-	-	-	-
[207]	1.4 2.9	30.0	30.0	g/kWh	-	-	-	200.0	750.0	ppm
[61]	2.0 3.1	0.1	0.6	g/kWh	-	-	-	-	-	-
[203]	2.0 4.0	0.2	2.0	g/kWh	-	-	-	-	-	-

Fig. 7. Engine exhaust emissions for lean combustion mode from various studies. Data unavailability is represented by a dash (–).

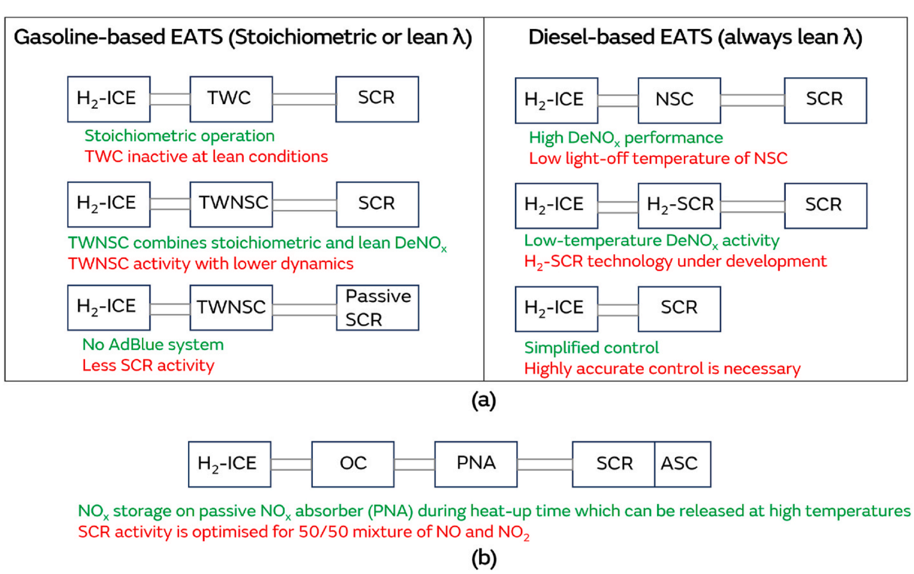

Fig. 8. EATS layout based on gasoline and diesel engines (a), adapted from Ref. [70], with permission from SAE International. Alternative EATS layout compatible with EU-7 regulations (b), reproduced from Ref. [225].

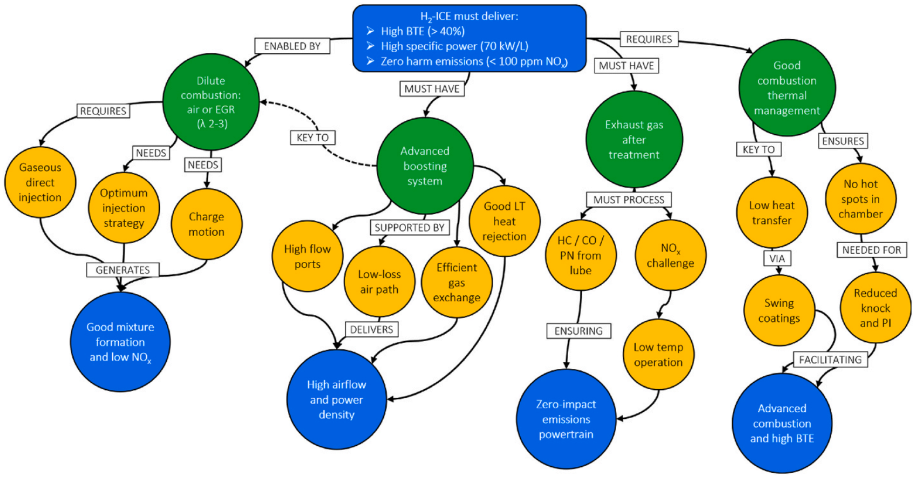

Fig. 9. Functional layout to achieve high thermal efficiency, specific power density, and low exhaust emissions.

7. Remarks and outlook

In recent years, the prospect of hydrogen as a future energy carrier has been identified by several automotive markets due to its low-to-zero carbon footprint which has led to the development of H_2-ICE for road transportation. H_2-ICE development has the potential to impact the heavy-duty, commercial, and light-duty sectors, with integration into hybrid powertrains of particular interest.

The development of onboard storage capability, supply, and distribution infrastructure is critical for the adoption of H_2-ICE technology for road transport and must be synchronised with vehicle development strategies. Early introduction of H_2-ICE architectures could include the relatively low-cost adaptations of conventional ICEs including dual-fuel capability and/or H_2-PFI development whilst addressing the key safety requirements of backfire from the intake manifold and crankcase gases. This generation of H_2-ICE powertrain can be developed without much emphasis on efficiency improvement [227] – simply displacing some of the fossil fuel-supplied energy with zero-carbon hydrogen. This near-term development can facilitate a smooth and continuous transition as the hydrogen supply chain and refuelling infrastructure develop over time. The second generation of the H_2-ICE powertrain will likely be a step forward in terms of efficiency and complexity and will thus facilitate the need for a combustion system optimised specifically for hydrogen direct injection which however might result in higher engine cost.

Therein lie several of the challenges for OEMs and researchers alike to fundamentally progress the H_2-ICE beyond the first generation offerings. The principal topics for research should include furthering our knowledge into mitigating knock and abnormal combustion, optimising in-cylinder mixing of hydrogen and air, in-cylinder heat transfer modelling and application of advanced materials, and expanding the capability of H_2-ICE 1D predictive combustion modelling.

It should be stated that this review paper is not intended to provide an exhaustive list of hardware optimisations needed for a highly efficient and ultra-clean H_2-ICE combustion system. The paper has, however, attempted to cover the key aspects of combustion systems and engine design that deserve developmental focus moving forward.

CRediT authorship contribution statement

Harsh Goyal: Writing – original draft, Methodology, Investigation, Data curation, Conceptualization. **Peter Jones:** Writing – review & editing, Resources, Project administration, Investigation, Conceptualization. **Abdullah Bajwa:** Writing – review & editing, Writing – original draft, Data curation. **Dom Parsons:** Writing – review & editing. **Sam Akehurst:** Writing – review & editing, Project administration, Funding acquisition. **Martin H. Davy:** Writing – review & editing, Project administration, Funding acquisition. **Felix CP. Leach:** Writing – review & editing, Project administration, Funding acquisition. **Stefania Esposito:** Writing – review & editing, Supervision, Project administration.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Abbreviations

ASC	Ammonia Slip Catalyst
BEV	Battery Electric Vehicle
BTE	Brake Thermal Efficiency
CCS	Carbon Capture and Storage
CCV	Crankcase Ventilation
CKS	Chrome Ceramic Coating
CMCV	Charge Motion Control Valve
CO	Carbon Monoxide
CO_2	Carbon Dioxide
DI	Direct Injection
EATS	Exhaust After-treatment Systems
EGR	Exhaust Gas Recirculation
EPA	Environmental Protection Agency
e-SC	Electric Supercharger
EU	European Union
FIE	Fuel Injection Equipment
FIP	Fuel Injection Pressure
GDI	Gasoline Direct Injection
H_2DI	Hydrogen Direct Injection
H_2-FCEV	Hydrogen Fuel Cell Electric Vehicle
H_2-ICE	Hydrogen Internal Combustion Engine
HC	Hydrocarbon
HE	Hydrogen Embrittlement
HRR	Heat Release Rate
HTHA	High-temperature Hydrogen Attack

ITE	Indicated Thermal Efficiency
IVC	Intake Valve Closing
JGC	Jet Guiding Caps
LIF	Laser Induced Fluorescence
LP	Low-pressure
MMO	Modified Microalgae Oil
NO_x	Nitrogen Oxides
N_2O	Nitrous Oxide
NSC	NO_x Storage Catalyst
OEM	Original Equipment Manufacturer
OH*	Electronically Excited Hydroxyl
PEO	Plasma Electrolytic Oxidation
PFI	Port Fuel Injection
PIV	Particle Image Velocimetry
PM	Particulate Matter
$PM_{2.5}$	Particulate Matter with 2.5 μm Diameter
PM_{10}	Particulate Matter with 10 μm Diameter
SCR	Selective Catalytic Reduction
TDC	Top Dead Center
TKE	Turbulent Kinetic Energy
TSC	Thermal Swing Coatings
TWC	Three-way Catalysts
TWNSC	Three-way NO_x Storage Catalysts
uH_2	Unburned Hydrogen
VGT	Variable-geometry Turbine
YSZ	Yttria-stablised Zirconia
ZDDP	Zincdialkyldithiophosphate

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